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PRESSURE DROP ACROSS AN ORIFICE IN PULSING GAS FLOWS

S.S. Hall, D.D.S. Liu and D.J. Patmore

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by

S.S. Hall*, D.D.S. Liu** and D.J. Patmore***

ABSTRACT

The pressure drop across an orifice as a function of time for pulsing gas flows is described. The operation of the CANMET hydrocracking pilot plant was analyzed either one or two compressors in parallel during the startup of a hydrocracking experiment prior to the injection of liquid.

The accuracy of calibration curves, which were measured using steady continuous flows for application in pulsating flows, was assessed.

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CHUTE DE PRESSION AU-DESSUS D'UN ORIFICE D'ÉCOULEMENT DE GAZ PULSÉS

par

S.S. Hall*, D.D.S. Liu** et D.J. Patmore***

RÉSUMÉ

L'utilisation de la chute de pression au-dessus d'un orifice comme fonction de temps de l'écoulement des gaz pulsés est décrite dans le présent rapport. L'exploitation d'une usine-pilote d'hydrocraquage a été étudiée à l'aide d'un ou deux compresseurs parallèles avant l'injection de liquide au cours de la mise en marche d'un essai d'hydrocraquage.

Lors de ces essais, on a également évalué la précision des courbes d'étalonnage, mesurées à l'aide d'écoulements en continue réguliers, pour leur utilisation possible avec les écoulements jaillissants.

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INTRODUCTION

Orifice meters are commonly used in steady flow measurements. They are also often used to monitor flowrates of pulsing flows. In many cases, the differential pressure or pressure drop (d/p) across the meter is a periodical function driven by piston compressor(s). The d/p variation could be large, but it may not show on the meter if a long rise time instrument is used. Therefore, it is important to ensure that all d/p values are within the range of its instrument in order to obtain an accurate measurement. If many d/p peaks exceed the range of the instrument, the reading (average value) could be inaccurate although the displayed reading could still be in the range of the meter.

This report describes a theoretical analysis of pressure drops across an orifice in pulsing flow. Examples are given for CANMET hydrocracking pilot plant operation using either one or two compressors in parallel.

This work was originally motivated by the following:

1. theoretical model is being developed to obtain rheological properties of liquid or slurry at elevated temperatures and under high pressure of hydrogen atmosphere by using data from pilot plant experiments (1). These data are:
   (1) the pressure drop (d/p) across the orifice and
   (2) the d/p from after the gas recycle pump to the bottom of the reactor in the CANMET hydrocracking pilot plant experiments. Therefore, a sub-model for the prediction of d/p across the orifice during the operation is required, and

2. the design of the CANMET coprocessing pilot plant (PP-3), the gas flow measurement system must be specified and instruments purchased (2). This model can be used to estimate design parameters and instrumental specifications.

GEOMETRY AND MONITORING INSTRUMENTS

This study is based on the geometry of the CANMET hydrocracking pilot plants (Fig. 1). The variable to be modelled is the pressure as a function of time between gas pump and orifice.

There are three d/p cells installed in the gas recirculation system to monitor the following pressure drops:

1. across the orifice (Fig. 2, D/P #1; from 0 to 100 in. of water),

2. from after the pump to the bottom or the reactor (Fig. 2, D/P #2; from 0 to 50 psi), and

3. across the compressor (Fig. 2, D/P #3; from 0 to 200 psi): the system head loss.

Experimental data from the d/p #1 cell will be used to compare with predictions.

Because the maximum d/p of the #1 d/p cell was limited to 100 in. of water, different size orifices were used for monitoring different flowrates. The flowrates versus pressure drops for four typical orifices were calibrated by using steady flows and are shown in Fig. 3. In the pilot plant operation, the recycle gas is a regular pulsing flow driven by piston compressor(s). Although traps were installed to reduce the pressure surging, it is expected that the d/p across the orifice would fluctuate severely. This is also the subject of this study.

THE MODEL

MASS BALANCE

Consider the volume from the compressor to the orifice (Fig. 3) contains two traps which reduce the pressure surge caused by the periodical compression. If the friction of the

pipe wall can be neglected, the pressure in the control volume is homogeneous. The mass balance gives:

Rate of Accumulation = (Input rate) - (Output rate) Eq 1

Since the input rate is given by the discharge of the compressor which is a piston pump, the input rate $[Q_{in}(t)]$ may be described by the product of a constant rate Q_{st} and a unit periodical function $F(t/t_s)$ (3):

$$Q_{in}(t) = Q_{st}\, F(t/t_s) \qquad \text{Eq 2}$$

where

$F(t/t_s) = 1 + L^{-1}\{(1/s)\tanh(s/2)\}$
Q_{st} = (Volume per stroke)/t_s
$L^{-1}\{\ \}$ = reversed Laplace transform operator
s = the Laplace transfer parameter
t = time
t_s = time for half stroke

For operations using multiple compressors in parallel, Eq. 2 becomes:

$$Q_{in}(t) = \sum[Q_{st,i}\, F(t_i/t_{s,i})]$$

$$= Q_{st,i} \sum F(t_i/t_{s,i}) \quad \text{[for identical pumps]} \qquad \text{Eq 2a}$$

where subscript i stands for the ith compressor.

The mass rate Q_{st} is related to the volumetric rate v_p by:

$$Q_{st} = v_p\, d_a \qquad \text{Eq 3}$$

where

v_p = piston volumetric discharge rate $[L^3/T]$
= (piston cross-sectional area) x (speed)
d_a = ambient gas density (ambient = outside of the control volume)

The v_p was assumed to be constant implying that the movement of compressor piston is linear with time. Under this assumption, the rate of gas input is a square-wave function of time. It is positive during the half cycle of the compressor's discharge mode. There is no input gas into the control volume in the intake half cycle of the compressor. The $F(t/t_s)$ versus time is illustrated graphically in Fig. 4.

The output rate with respect to the control volume can be evaluated at the orifice:

$$Q_{out}(t) = U_o(t)A_o d_o \qquad \text{Eq 4}$$

where

A_o = orifice opening [L^3]
d_o = density [M/L^3]
$U_o(t)$ = gas velocity [L/T]

The subscript "o" stands for "at orifice".

When the system pressure is high,

$$d_o \simeq d_{trap} = zPM/RT \qquad \text{Eq 5}$$

where

M = molecular weight [M/M-mole]
P = pressure in the trap [Force/L^2]
R = gas constant
T = temperature
z = compressibility

Substituting Eq. 2 and 4 into Eq. 1:

$$d(V\,d)/dt = Q_{in}(t) - U_o A_o d_o \qquad \text{Eq 6}$$

or combining with Eq. 5 to give:

$$V\, dP/dt = v_p\, F(t/t_s)\, P_a - U_o(t) A_o\, P \qquad \text{Eq 7}$$

where

P_a = ambient pressure, i.e., pressure outside or control volume (13.9 MPa)
P = pressure in the control volume
V = volume of the control volume
= total volume of traps if volume of lines is neglected

The gas velocity at orifice, $U_o(t)$, can be expressed by (4,5):

$$U_o = [C_o/(1 - b^4)^{1/2}][2g_c(P - P_a)/d_o]^{1/2} \qquad \text{Eq 8}$$

C_o is the orifice coefficient and must be experimentally determined for the orifice used; b is the ratio of orifice diameter to pipe diameter and is usually very small. Thus $b^4 = 0$ can be assumed. P_a is a variable during the hydrocracking operation. Because the liquid feed was not yet considered in the system, P_a = 13.9 MPa can be assumed.

There is absolutely no question about the adequacy of Eq. 8 in transient applications. This equation is well known and has been extensively reviewed and tested including fast transient applications (4-6). It will be shown that the transients considered here are very slow thus the applicability of Eq. 8 is warranted.

In the remainder of this paper, d/p refers to $P - P_a$.

When the S.I. convention is used, Eq. 8 can be be written as:

$$U_o = C_o\, [(2RT/zM)(P - P_a)/P]^{1/2} \qquad \text{Eq 9}$$

In the derivation of Eq. 9, Eq. 5 has been used.

Inserting Eq. 9 into Eq. 7 results in:

$$dP/dt = v_p F(t/t_s) P_a/V - (A_o C_o/V)[(2RT/zM)P(P-P_a)]^{1/2} \quad \text{Eq 10}$$

Equation 10 describes the rate of pressure variation in the control volume (Fig. 2). It is an ordinary differential equation (ODE) of the following general form (8):

$$dP/dt = G(t,P) \quad \text{Eq 11}$$

where G(t,P) stands for the right hand side expression of Eq. 10.

Numerical methods are available for solving Eq. 11. The Runge-Kutta method was adopted (8-9).

The effect of the instrument damping on the $P - P_a$ signal is not included in this study.

COMPRESSOR PUMPING VOLUME

The piston compressors are assumed to have a constant stroke volume. The average volumetric rate is controlled by the frequency of strokes.

The stroke volume of a compressor in PP-2 was estimated to be 8.56 cm^3 by the following method (10):

(1) the frequency of strokes at a given d/p setting was measured,
(2) the average volumetric rate was assumed to be equal to that of the steady (continuous) flow which would give the same d/p reading,
(3) the stroke volume is assumed to be the value of result (2) divided by the result of (1).

This value was determined for one condition only and was used for all calculations. Because two compressors in PP-2 are identical, 8.56 cm^3 is assumed to be the stroke volume for both.

FUNCTION OF THE CHECK-VALVE

A check-valve is installed vertically in the gas line before the junction with the liquid feed line (Fig. 2). This is to prevent the pitch flowing back to the gas line where the temperature is usually below the freezing point. The check-valve is simply a steel ball-seat design. It is based on the gravitational force of the steel ball against the upstream gas pressure. The ball is made of stainless steel and is 5 mm in diameter and the diameter of the seat open is 3 mm. A simple model or its operation, assuming the valve is an on/off valve, is described in Appendix A.

NUMERICAL METHOD AND SIMULATION

ORDER OF RUNGE-KUTTA NUMERICAL METHOD

Details or the Runge-Kutta method for solving the ODE Eq. 11 can be found in textbooks (8,9). Depending on the degree of accuracy required, the 2nd, 3rd or 4th order of the Runge-Kutta method can be used. The "order" reflects the order of accuracy compared with the order ot Taylor expansions. The higher order method would, however, require a longer computing time.

The results of Runge-Kutta calculations are predictions for the pressure in the control volume as a function of time. Figure 4 shows a portion of the result for the operation of PP-2 at 45.3 L/h (1.6 ft^3/h) calculated by using the 2nd order method for stepsizes of 0.088 s and 0.0088 s. The on/off function [$F(t/t_s)$] of the compressor is indicated by the 1/0 square wave on the same diagram. It matches the observed pressure history well.

To choose the order of the Runge-Kutta method, a series of tests was carried out. Some results are given in Fig. 5 compares:

(1) 2nd order Runge-Kutta method using a stepsize of 0.0125 s (20

steps per half stroke), and
(2) 4th order method using a stepsize of 0.00125 s (200 strokes per half stroke).

The agreement between these two cases is good. It indicates that the 2nd order Runge-kutta method using a relatively large step would produce satisfactory results and was thus adopted in this work.

NUMERICAL SIMULATION

Calculations were carried out for pilot plant operation using either one or two compressors in parallel. Results are presented below:

I. Single Compressor Operation:

Two calculations were carried out for operation when only one compressor was employed. The variables examined were the trap volume (PP-1: 4.5L, PP-2: 6L) (Fig. 6) and the average volumetric flowrate (Fig. 7).

Figure 6 shows the first 2.5 s prediction examining the effect of trap volume (6 L versus 4.5 L) at the same pumping rate of 45.3 L/h (1.6 ft^3/h) on pressure drop. The larger trap volume would produce smaller d/p fluctuations.

Figure 7 compares results for PP-2 operation at 56.6 L/h and 45.3 L/h pumping rates. Because a higher pumping frequency was required to acnieve a higher average volumetric rate, the system pressure increased faster for the higher rate in the initial pumping stage.

The effect of the check-valve on the rate of pressure buildup was examined for a 1 psi check-valve (Appendix A). Because additional head was required to lift the check-valve's ball, it delays the pressure buildup at the initial stage (Fig. 8). The steady state d/p across the orifice would not be affected.

The calculation rapidly "converged", i.e., d/p peaks become repetitive in a few seconds. This is the "steady state" solution and is equivalent to the normal operating condition. A short time to establish equilibrium is expected when a small trap volume is used.

II. Dual Compressor Operation:

Three calculations were carried operations when two compressors were used simultaneously.

Variables examined in these tests were again the trap volume and the total average volumetric flowrate. The interaction (Eq. 2b) between compressors resulted in complicated patterns of d/p as a function of time. To achieve a desired mean flowrate, various ratios of pumping rates can be used. Therefore, the effect of compressor flowrate ratios was also examined.

1. Effect of Compressor Flowrate Ratios:

Three ratios of compressor flowrates (about 1/1, 1/1.5 and 1/2.5) at a total average flowrate of ~99 L/h were tested. The results are shown in Fig. 9 to 11.

Figure 9 shows results for the pumping rate ratio ~1: V_1 = 48.68 L/h and V_2 = 50.09 L/h. The difference 1.41 L/h (= 0.392 cm^3/s) between pumping rates of two compressors generates a 21.85 s [= 8.56 (cm^3/stroke)/0.392 (cm^3/s)] period of interference in the d/p history, i.e., the time from one major node to another is 21.85 s. Similar interference patterns were often observed by operators in pilot plant operation when two compressors in parallel were used.

Figure 10 shows the results for V_1 = 39.34 L/h and V_2 = 59.15 L/h (pumping ratio ~1/1.5). The difference between pumping rates was 19.81 L/h which produces a series of 1.55 s interference cycles. Each contains 3 d/p peaks.

Results for the pumping ratio ~2.5: V_1= 28.02 L/h and V_2 = 71.03 L/h are shown in Fig. 11. Each cycle contains about five peaks. The interference pattern was rather complicated because the frequencies are close together.

2. High Volumetric Rate Operation:

The effect or volumetric rate on the d/p history was examined for the operation of two compressors simultaneously with a total flowrate of 148.8 L/h at a pumping ratio ot ~1 (V_{1b} = 73.3 L/h and V_{2b} = 75.3 L/h).

Figure 12 displays a portion of the result. The d/p interference pattern is similar to Fig. 9 for a smaller flow rate. However, the values of d/p are now all above zero.

3. Effect of trap volume:

The effect of trap volume on the pressure drop across the orifice in multi-compressor operations is similar to the effect in single compressor systems. The fluctuation ot the d/p is larger when a smaller trap volume is used. Figure 13 shows the predicted pressure drop for a trap ot 4.5 L (PP-1). Significantly larger variations in d/p than those shown in Fig. 9 for a 6L trap (PP-2) were observed.

DISCUSSION

The following two pilot plant experiences can now be explained:

1. It was found that the damping ot newly purchased d/p cell for orifice meters must be adjusted to damp out the fluctuation. Without this adjustment, the "green pen" (d/p cell #2) will fluctuate widely creating difficulty in monitoring the flowrate. It is clear from the results of this study that the d/p across the orifice in the pilot plant operation does, indeed, fluctuate widely (Fig. 5-13). The rise time of d/p cells is usually set by the

manufacturer for continuous flow applications. It is short in order to detect any abnormal fluctuations. Thus the adjustment of rise time for newly purchased d/p cells is needed for average flowrate measurements in pulsing flows.

2. In some runs with high gas rate using two compressors at slightly different rates, the band of the "green pen" output widened and narrowed in turns. These were the interference patterns similar to those shown in Fig. 9 and 10.

This study verifies that the variation of d/p across the orifice is large in pilot plant operation. It can be improved if a larger trap volume is used.

When the damping of the orifice meter was adjusted to obtain a sufficient risetime, it was expected that the reading would be close to the temporal mean of the d/p values at steady state operation,

$$\langle P \rangle = (1/t_a) \int_0^{t_a} P \, dt \qquad \text{with} \qquad t_a >> t_s$$

It is therefore interesting to compare the calculated <P> values with results of calibration using steady continuous flows.

Although the above calculations were based on z = 1.0 whereas under 13.9 MPa, z = 1.1 for hydrogen, calculations were not repeated for z = 1.1 because z is a constant in Eq. 10 and the calibration curves can be converted for z = 1.0 (Appendix B) to have a realistic comparison with the predicted values. The results of converted z = 1 equivalent calibrations are plotted in Fig. 14 along with predicted temporal mean values for four orifices. The agreement is excellent for d/p >30 in. of water. Below this, the modelled d/p values are lower because the pressure upstream of the orifice was completely discharged in some pump intaking periods (Fig. 9 and 13).

CONCLUSIONS

The pressure drop across an orifice in pulsing gas

flows at high pressure was modelled. Predictions were made for pilot plant operation before injecting the liquid. Predicted temporal mean d/p values for four orifices at various pumping conditions agree well with values derived from calibration curves determined by continuous flows.

Good agreement between predicted and calibrated values was found for d/p >30 in. of water in either single-compressor or multi-compressor operation. During pilot plant operation an appropriate orifice size is always chosen to keep the d/p between 40 and 100 in. of water. The calibration curves, which were determined by using continuous flows, are therefore adequate for pilot plant application (11).

Because the range of the d/p cell is limited to <100 in. of water, it requires an orifice size large enough to maintain the pressure drop within this range. For the given stroke volume of 8.56 cm^3, the d/p will always fluctuate widely in any pilot plant operation. Linear gas velocity through the preheater is therefore a forced periodical flow. Its effect on hydrodynamics at the inlet of the hydrocracking pilot plant reactor is not known and would be very interesting for future study.

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APPENDIX A

FUNCTION OF THE CHECK-VALVE

A check-valve is mounted vertically in the gas line before the junction with the liquid feed line (Fig. 2). This is to prevent pitch flowing back to the gas line where the temperature is usually much lower than the freezing point. The check-valve is simply a steel ball-seat design. The ball is 5 mm in diameter and the diameter of the seat open is 3 mm. A simple model was used to examine the effect of this valve on the orifice pressure drop assuming the function of a check-valve is an on/off valve at 1 psi . The effect was observed to be a delay in the buildup of pressure in the traps (Fig. 8).

The choice of 1 psi was arbitrary for illustrative purposes. It can be quantified and is described in this appendix. It will be shown that the effect of the check-valve on the d/p across the orifice in "steady" pilot plant operation is negligible.

There will be no gas flow until the pressure in the trap increases to a level sufficient to lift the steel ball from its seat. Once the check-valve is opened the pressure drop across the orifice will increase to the same value as if there were no check-valve. Of course the pressure in the trap will have increased by the amount required to open the check-valve. A force balance was made on the valve to obtain a rough estimate of the opening pressure.

Force up = Force down

$$(\text{Pressure})(\text{Projected area}) = (\text{Mass})(g/g_c)$$

Assuming the steel ball density is 8500 kg/m^3 (stainless steel), the pressure required to just balance the gravity force would be 0.11 psi (3.10 in. of water). Additional pressure is needed to lift the ball from the seat and can be calculated from:

Pressure to lift = (Density)(Height lifted)(g/g_c)

To lift the ball one full diameter (5 mm) would require 0.06 psi pressure (1.67 in. of water). Roughly, a pressure increase of about 0.2 psi (~0.11 + 0.06) would be sufficient to open the valve. Considering that ambient pressure is 2000 psi, the increase (0.2 psi) as a result of the check-valve is negligible.

APPENDIX B

EXPECTED PRESSURE DROP FOR z = 1

The calibration curves given in Fig. 2 follow the general equation:

$$Q = k\,[(P - P_a) / d]^{0.5} \qquad \text{Eq B1}$$

The density d is given by:

$$d = 1 / \tilde{v} \qquad \text{Eq B2}$$

where the specific volume is given by:

$$\tilde{v} = zRT / P_a - zRT / P \qquad \text{Eq B3}$$

Substituting Eq. A2 and A3 into Eq. A1, the following relation is obtained:

$$P - P_a = (Q^2\, P_a / k^2 RT) / z \qquad \text{Eq B4}$$

For calibration curves, z = 1.1 was used (Fig. 2). In order to generate a set of corresponding curves for z = 1.0 to compare with results of model predictions in which a value of z = 1.0 was used, values of d/p in Fig. 2 must be multiplied by 1.1 for all values of Q. The results are shown by solid lines in Fig. 15. They agree well with predicted temporal mean values.

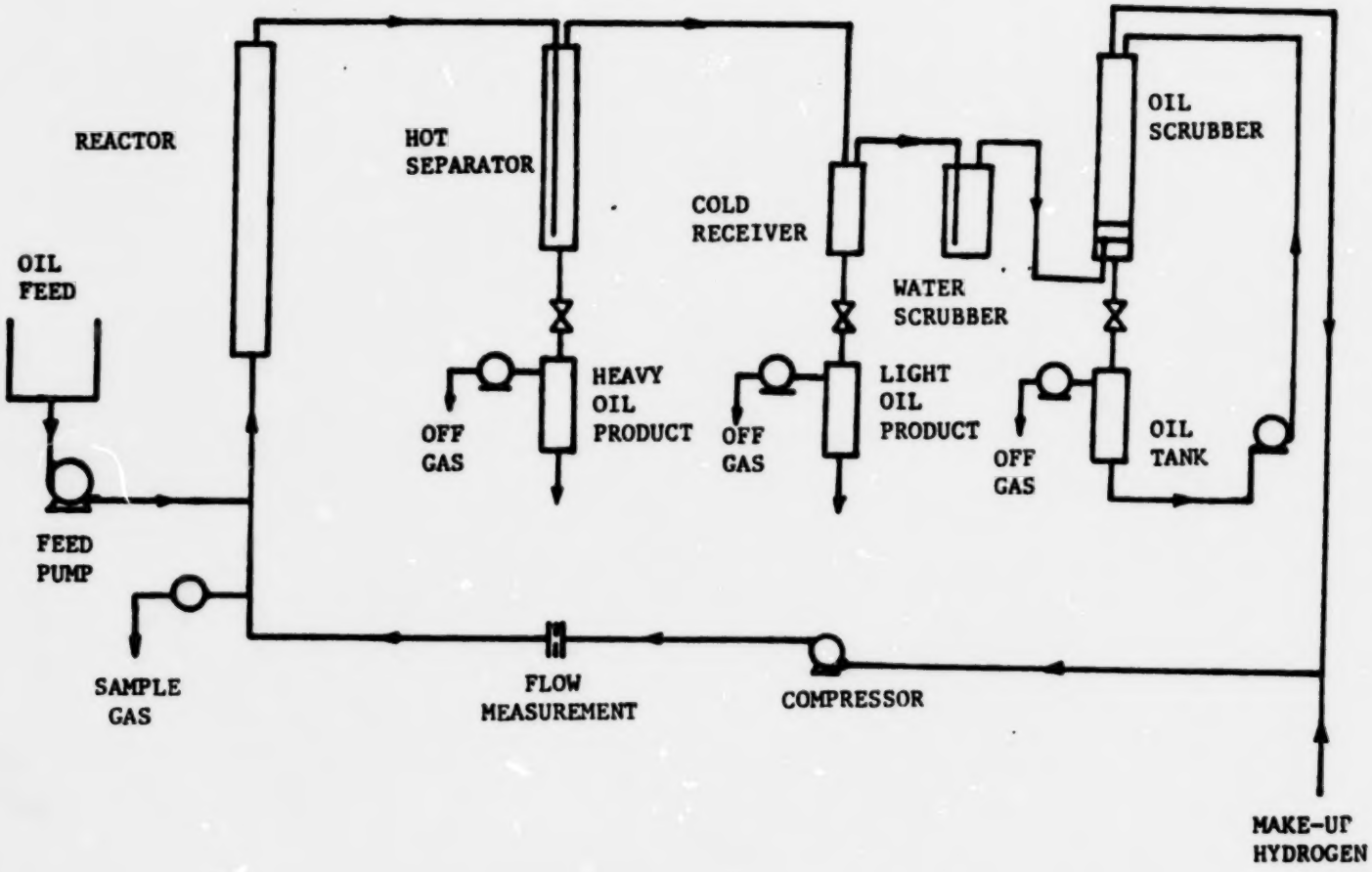

Fig. 1 - Schematic of CANMET hydrocracking pilot plant

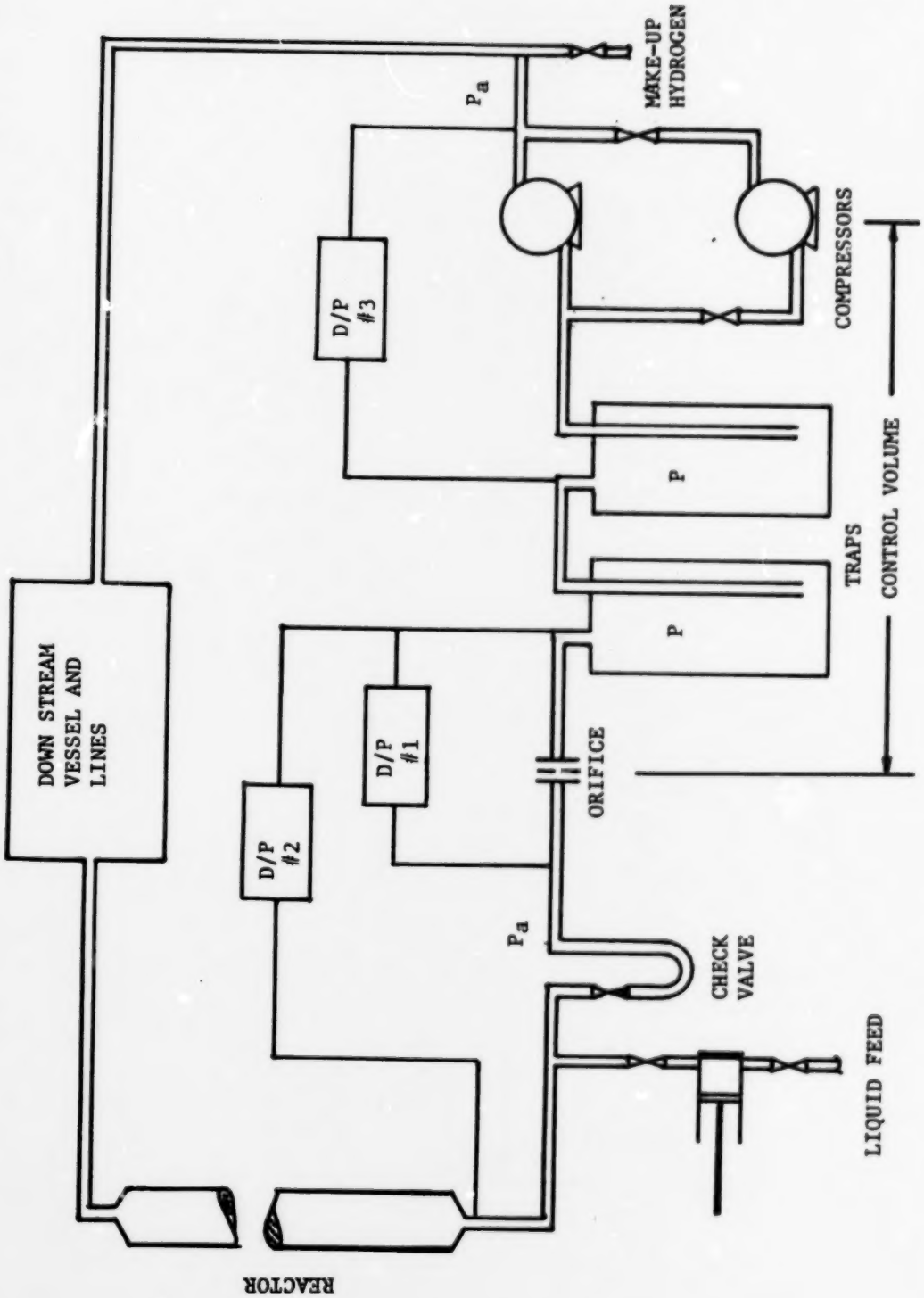

Fig. 2 - Schematic of locations of d/p cells

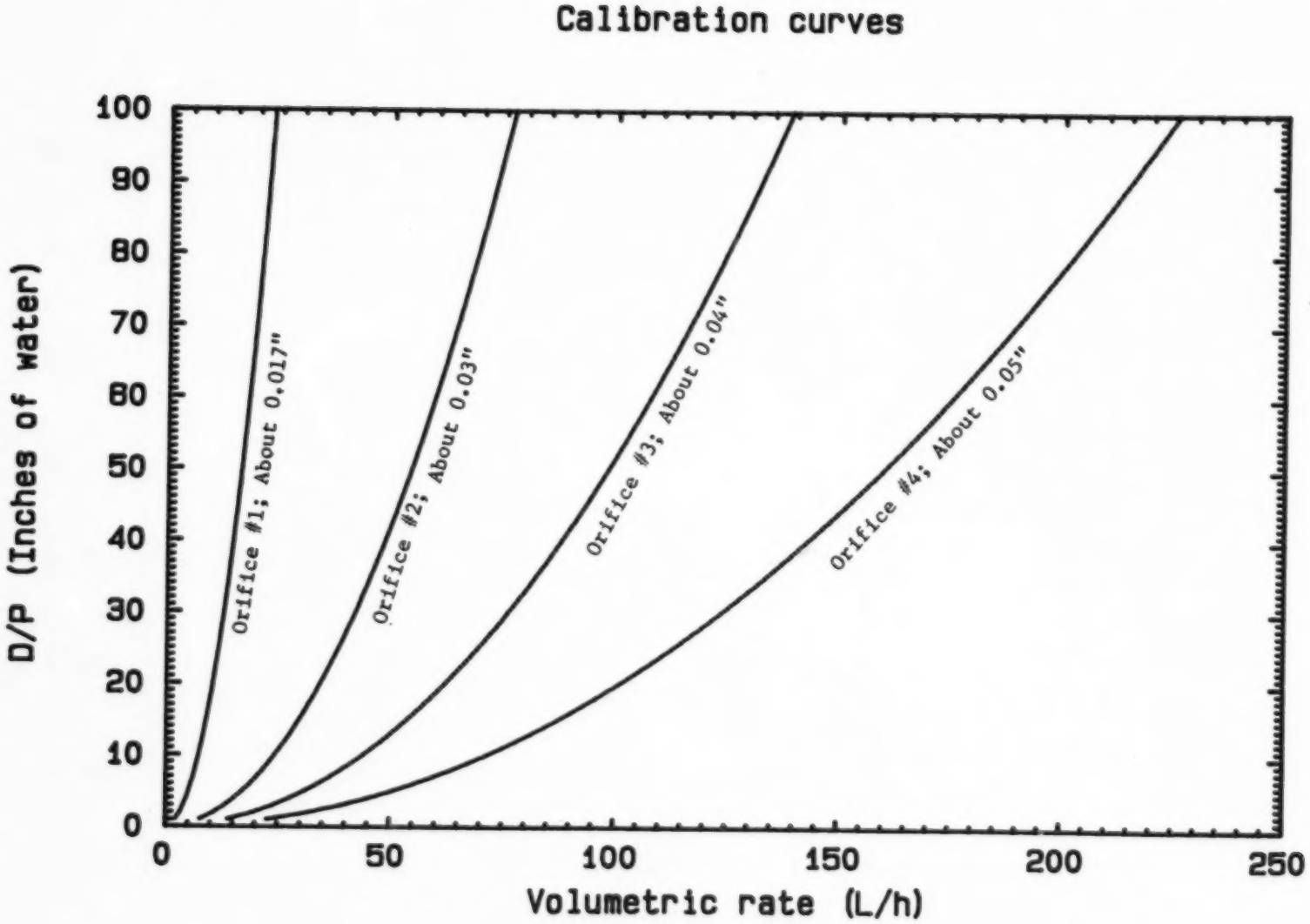

Fig. 3 - Differential pressure across four orifices at 13.9 MPa versus continuous hydrogen flow rates

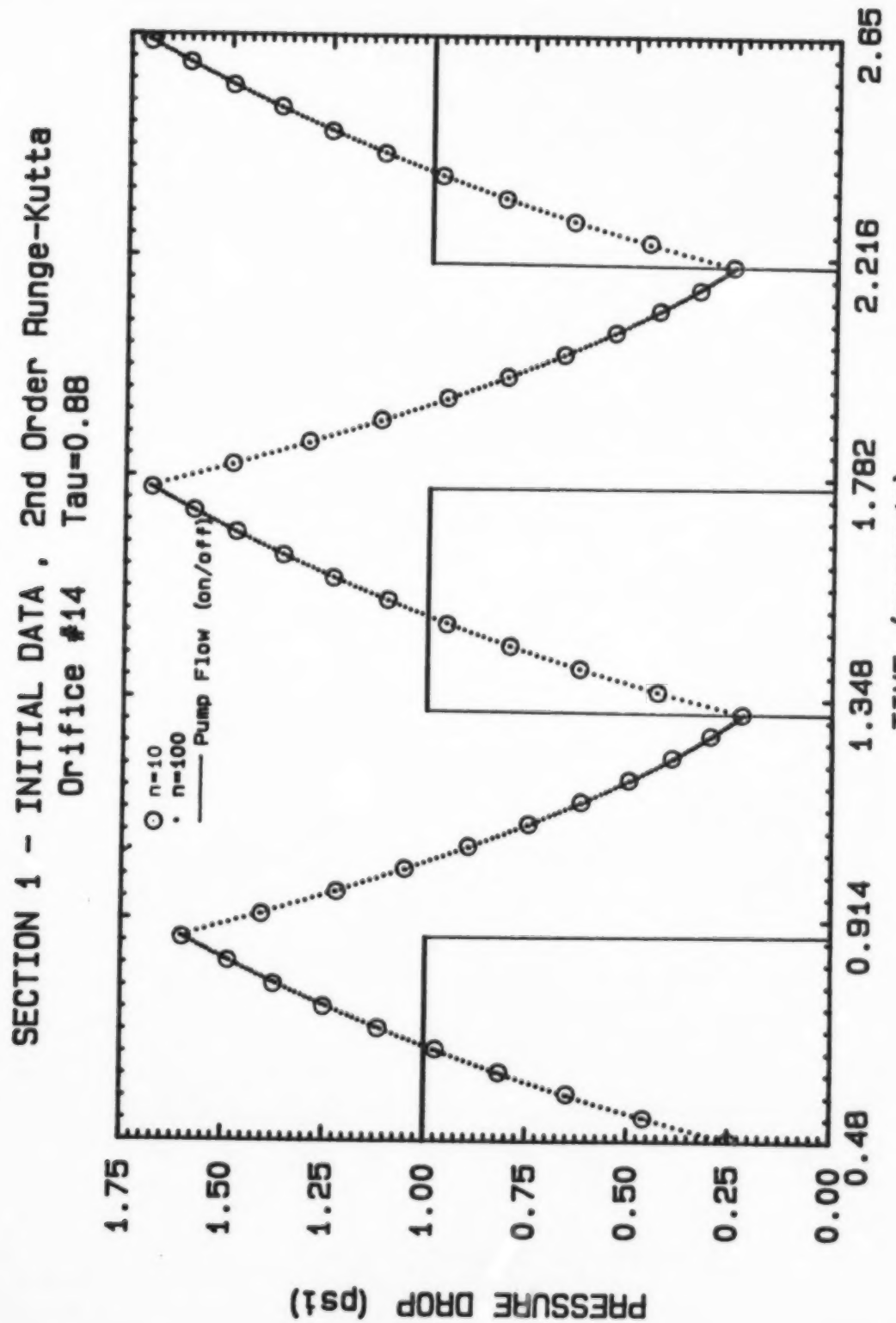

Fig. 4 - Comparison of pressure drop history with the functioning of compressor

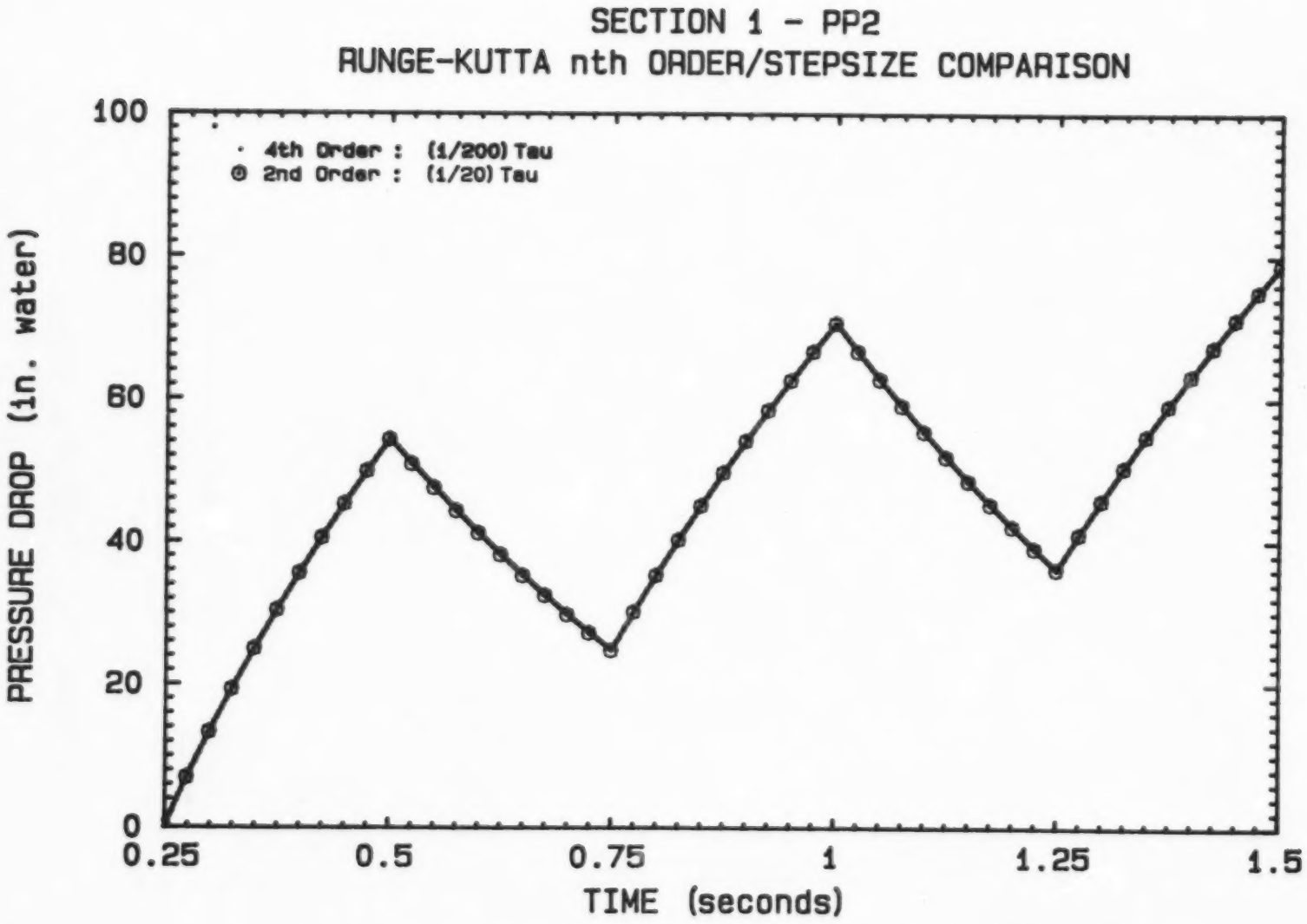

Fig. 5 - Comparison of 2nd and 4th Runge-Kutta methods (t_s = 0.25 s)

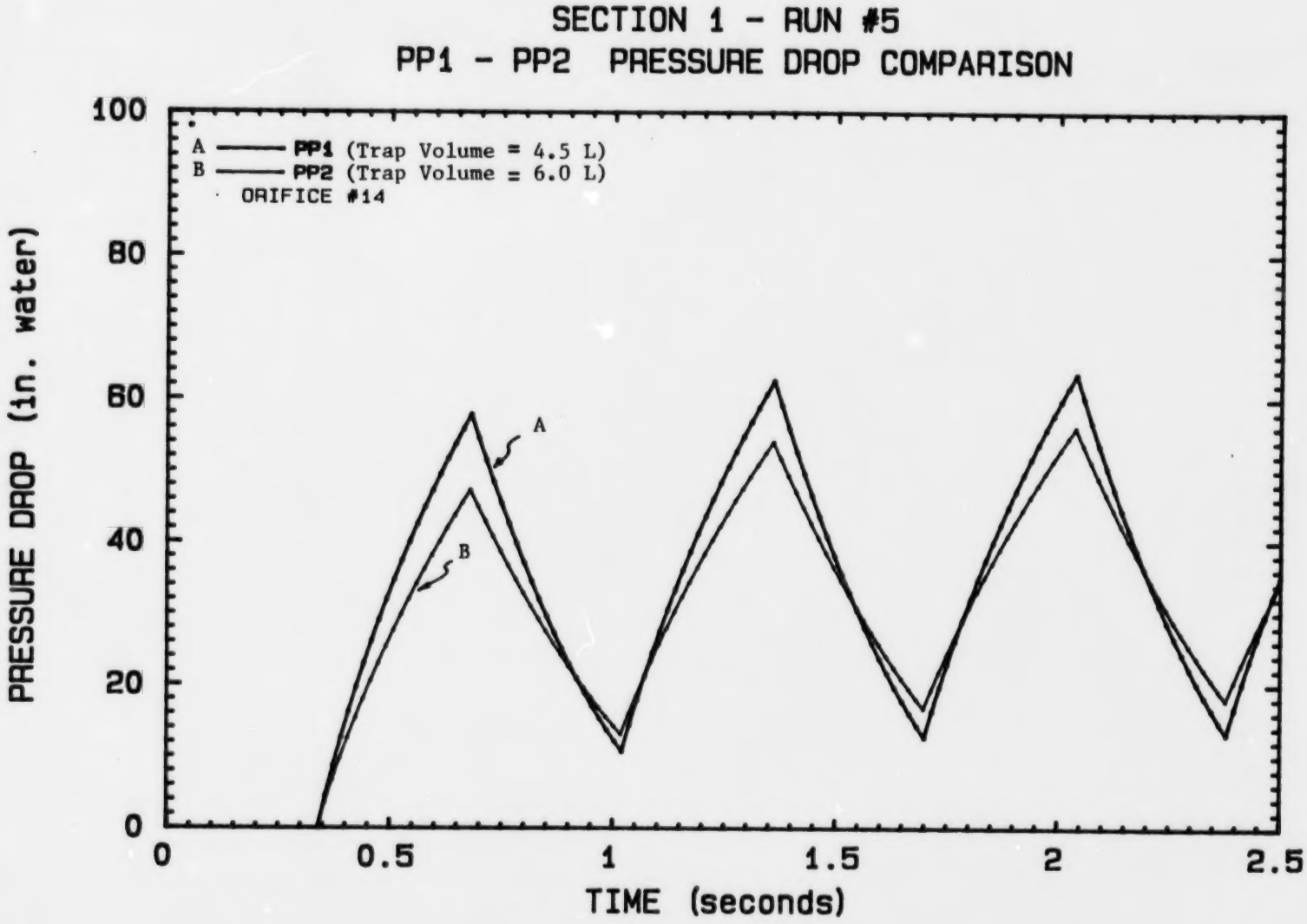

Fig. 6 - Effect of trap volume on the pressure drop across the orifice using single compressor

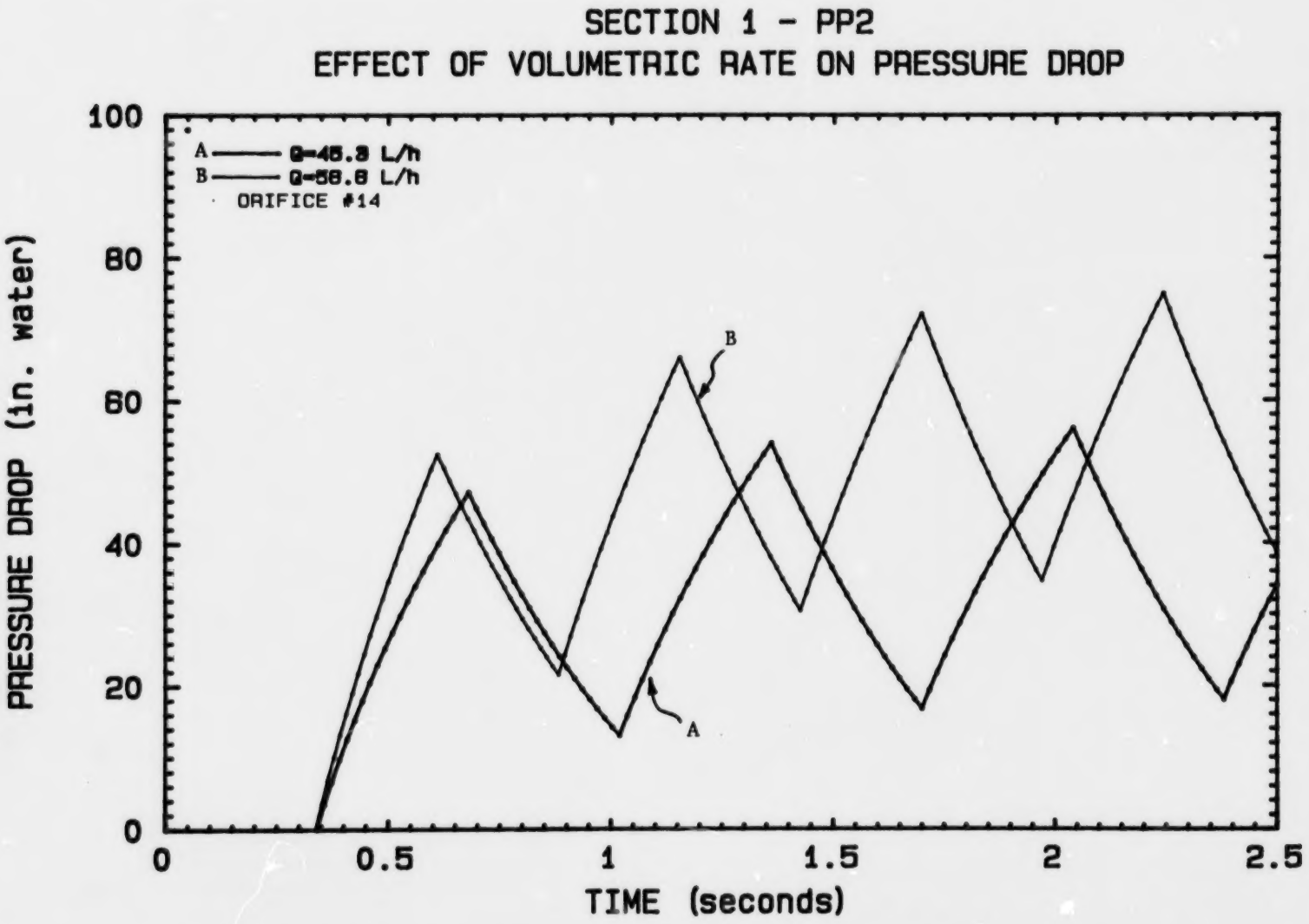

Fig. 7 - Effect or flowrate on the pressure drop across the orifice in the operation using single compressor

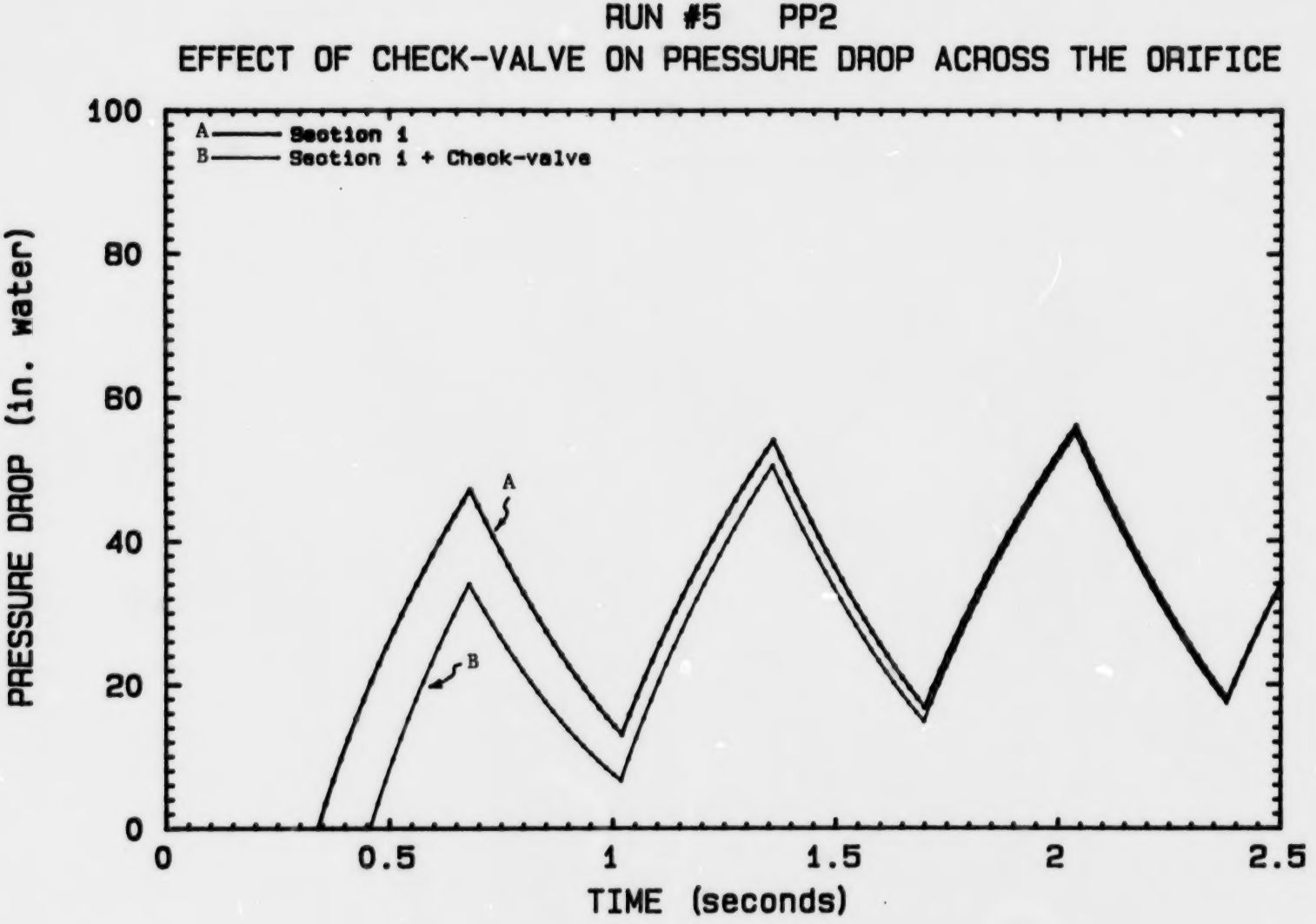

Fig. 8 - Effect of check valve on the pressure drop across the orifice using single compressor

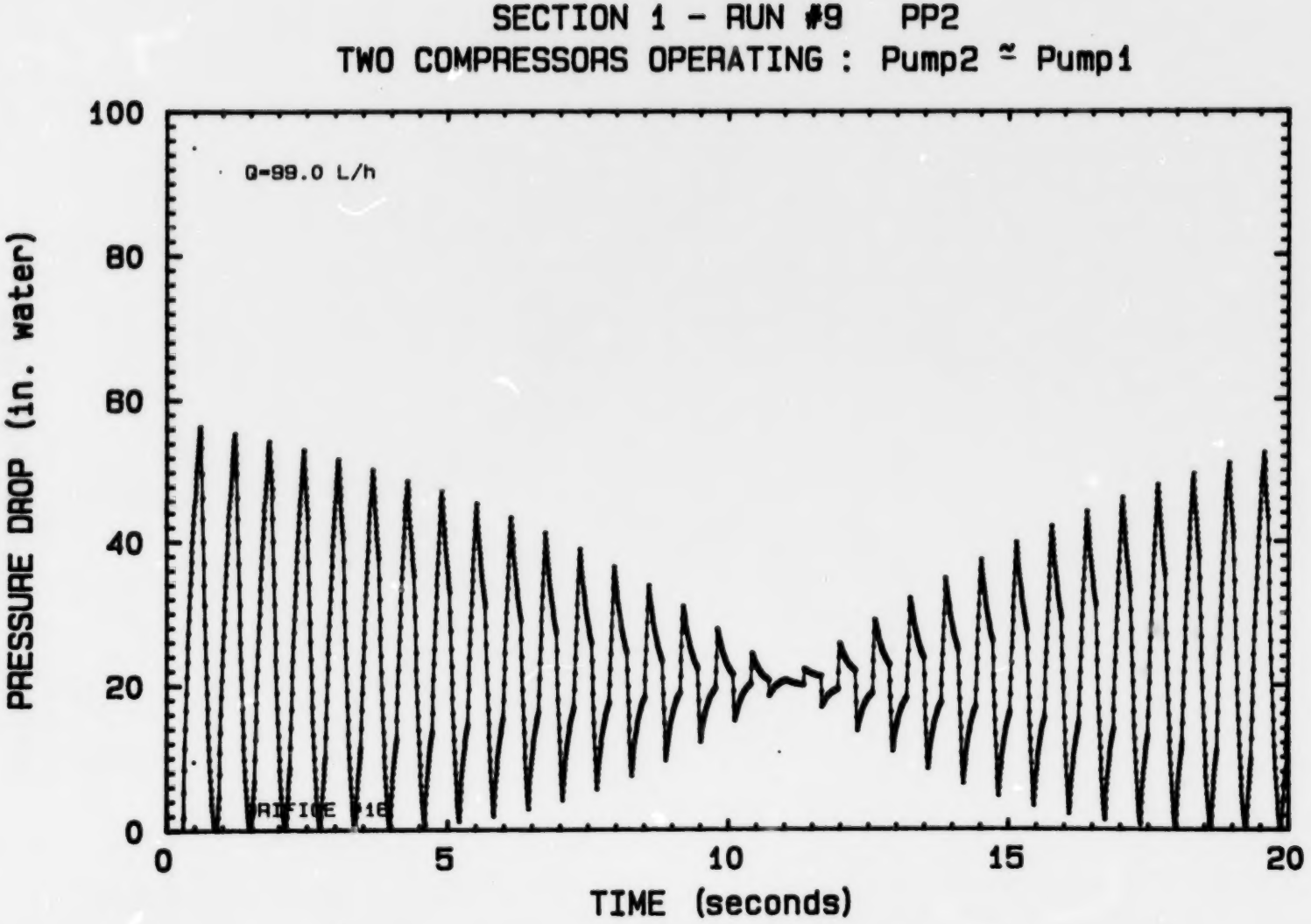

Fig. 9 - Pressure drop across the orifice as a function of time using two compressors at 99 L/h and pumping ratio 1/1 (Trap volume = 6 L)

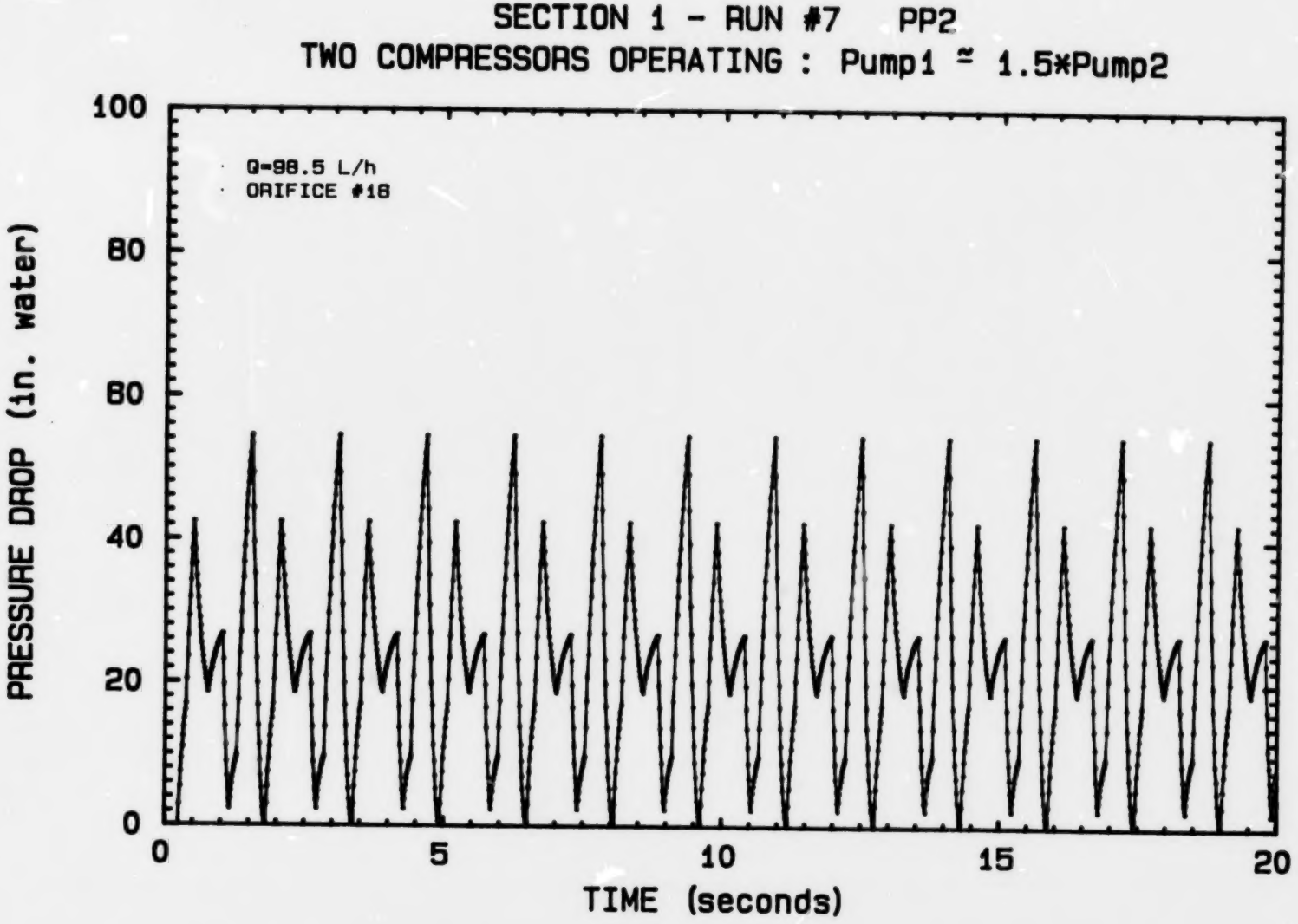

Fig. 10 - Pressure drop across the orifice as a function of time using two compressors at 98.5 L/h and pumping ratio 1/1.5 (Trap volume = 6 L)

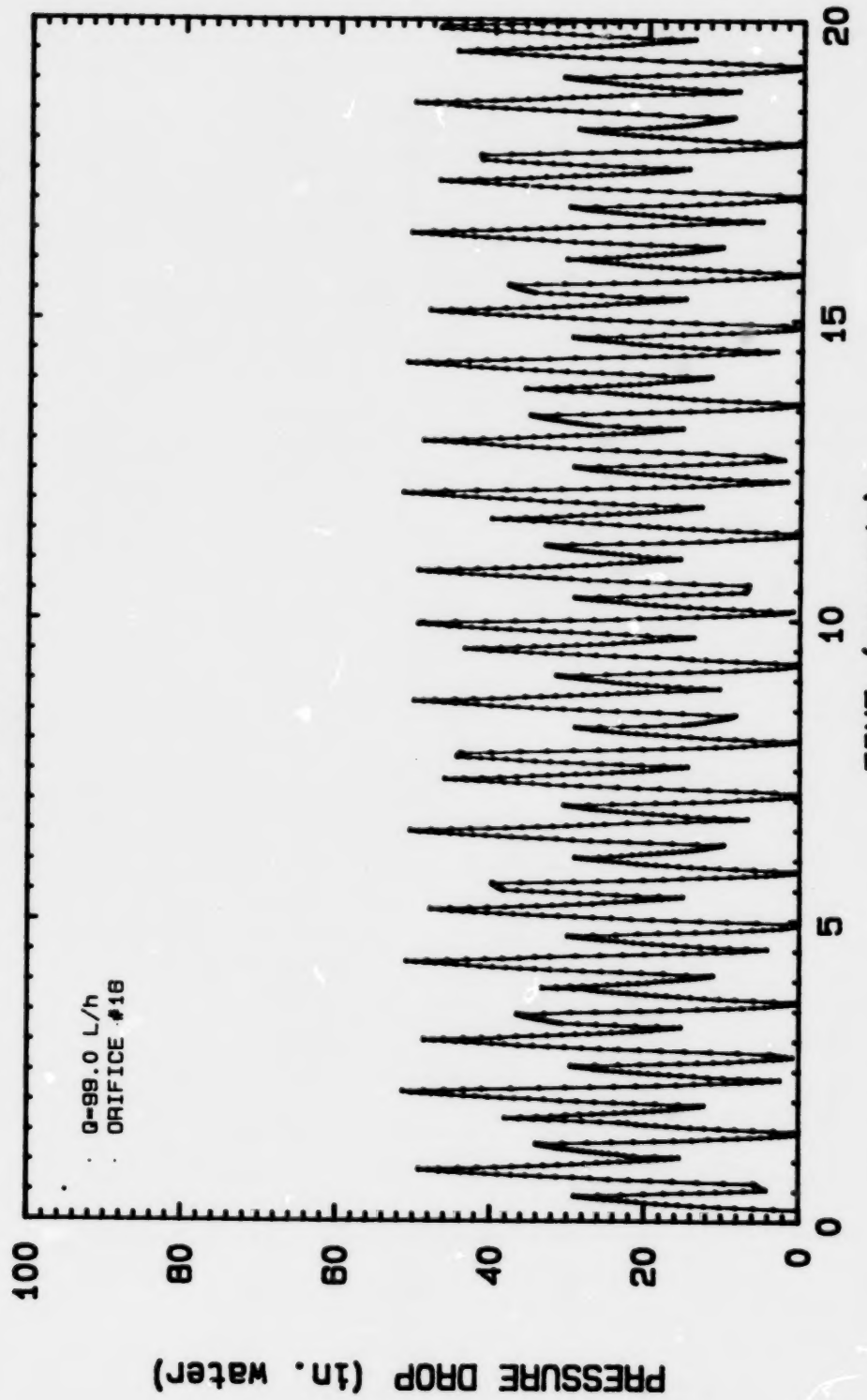

Fig. 11 - Pressure drop across the orifice as a function of time using two compressors at 99 L/h and pumping ratio 1/2.5 (Trap volume = 6 L)

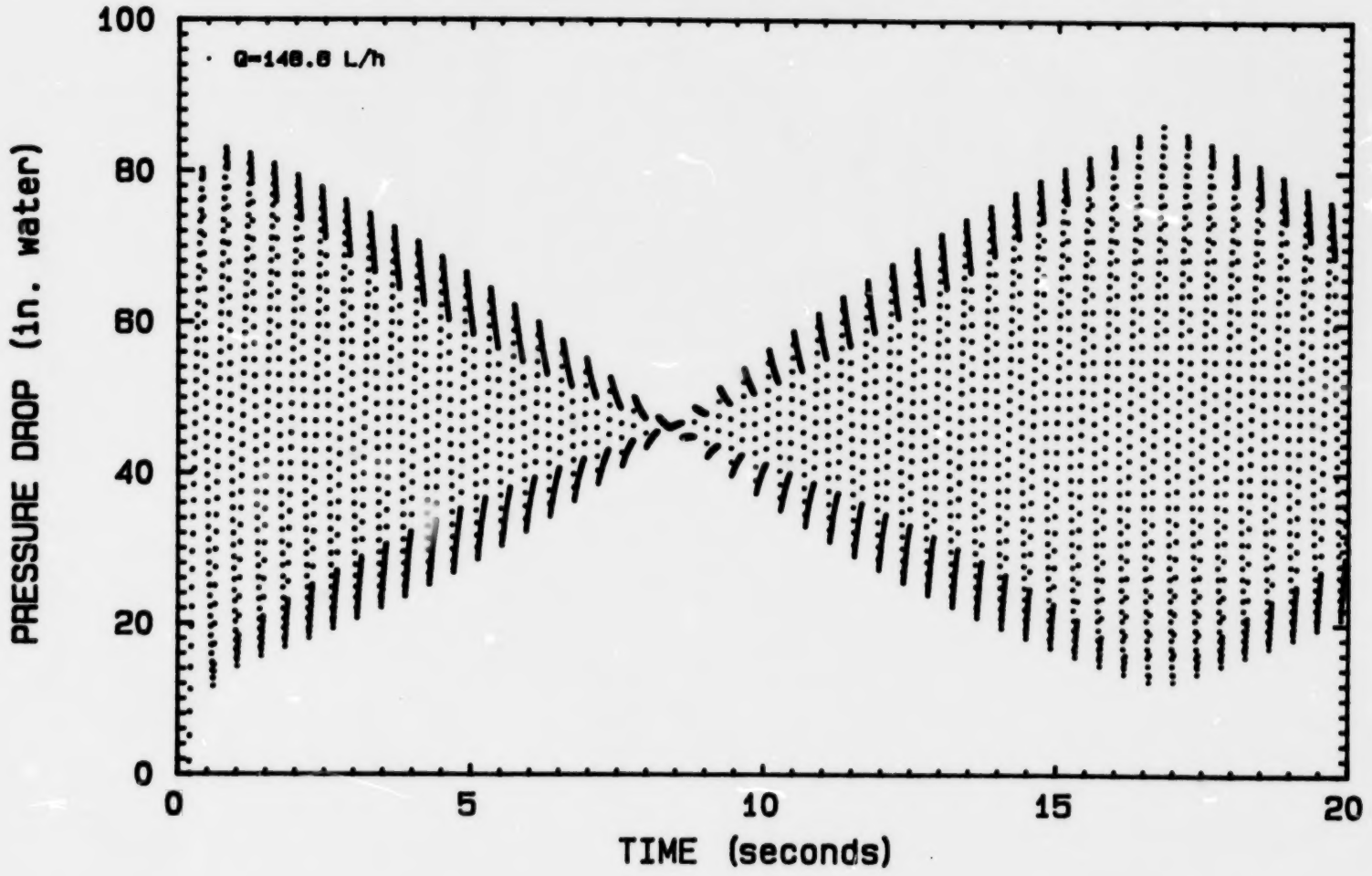

Fig. 12 - Pressure drop across the orifice as a function of time using two compressors at 148.8 L/h and pumping ratio 1/1 (Trap volume = 6 L)

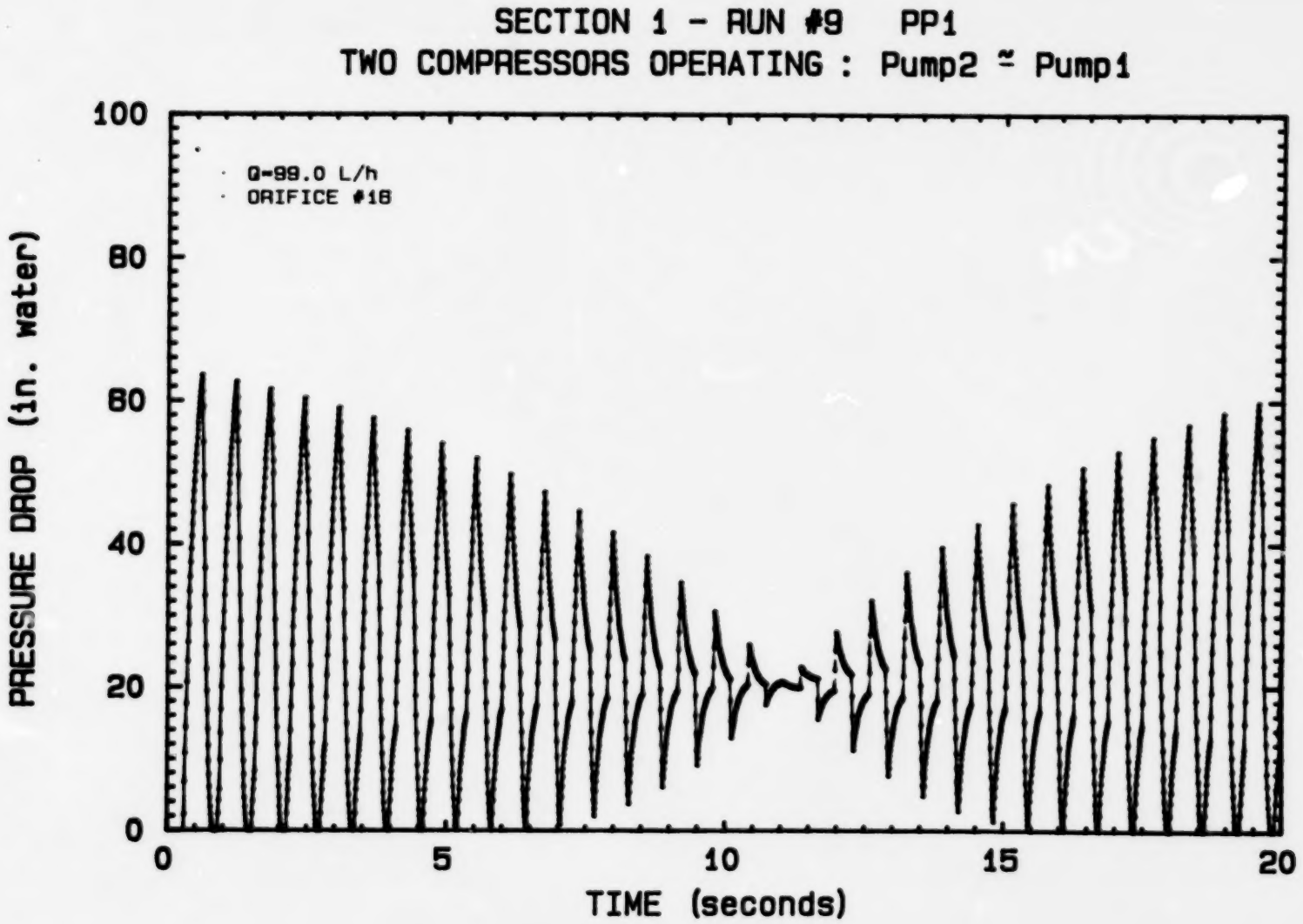

Fig. 13 - Pressure drop across the orifice as a function of time using two compressors at 99 L/h and pumping ratio of 1/1 (Trap volume = 4.5 L)

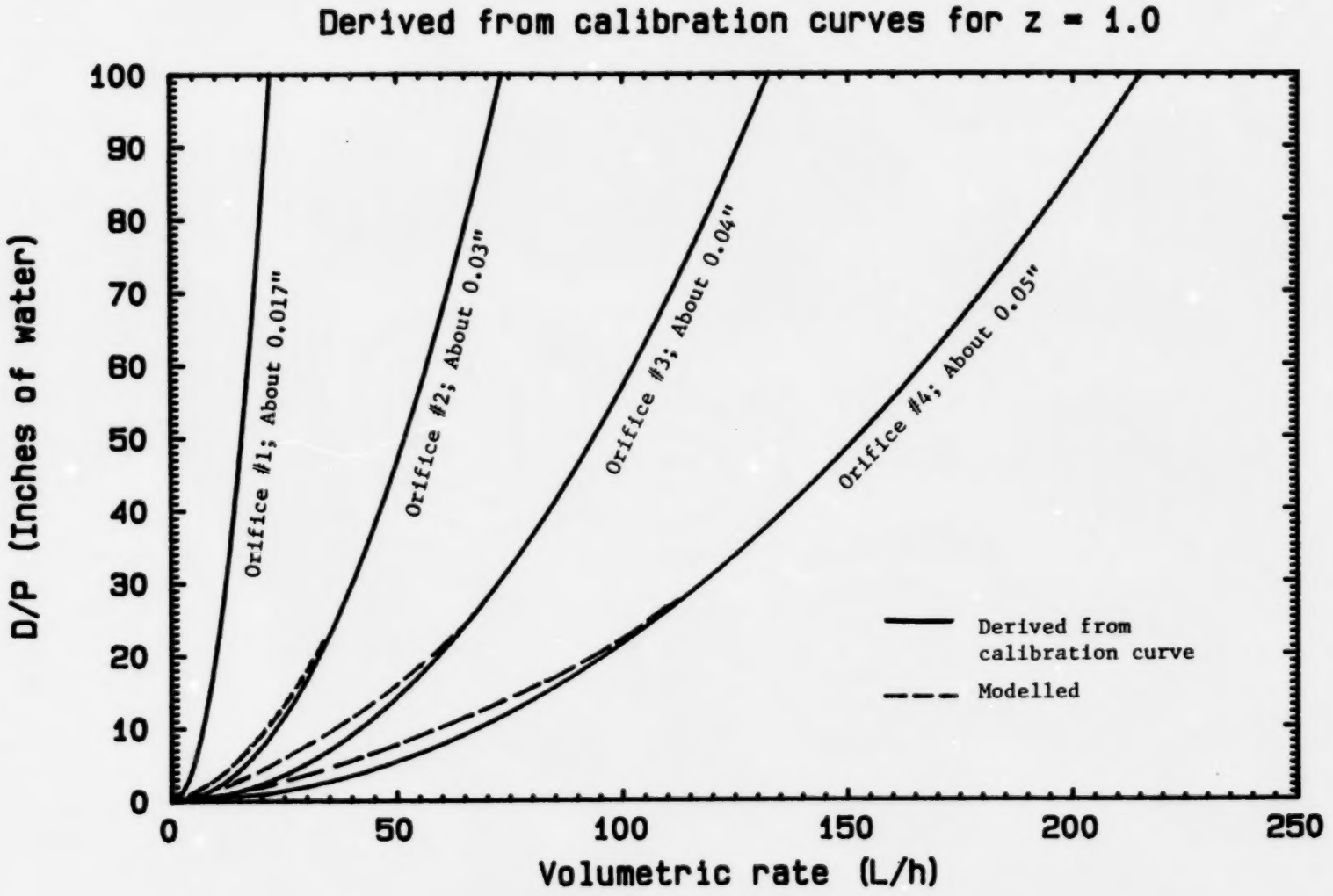

Fig. 14 - Comparison of mean values of pressure drop across orifices predicted for pulsing flows with values for continuous flows